



Calculating Nozzle Side Loads using Acceleration Measurements of Test-Based Models

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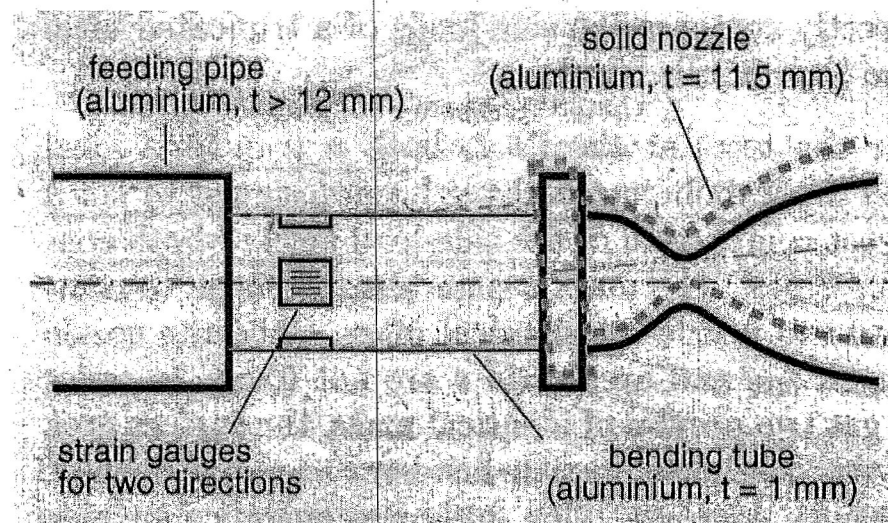
Motivation of Paper

1. Examine applicability and issues of using entirely modal test-based “pseudo-finite element models” for dynamic analysis.
2. Discuss methodology for “inverse-force determination” using operational test acceleration measurements.
3. Apply techniques to important Rocket Engine Nozzle Sideloads phenomena.



Present Side Loads Research at NASA/MSFC

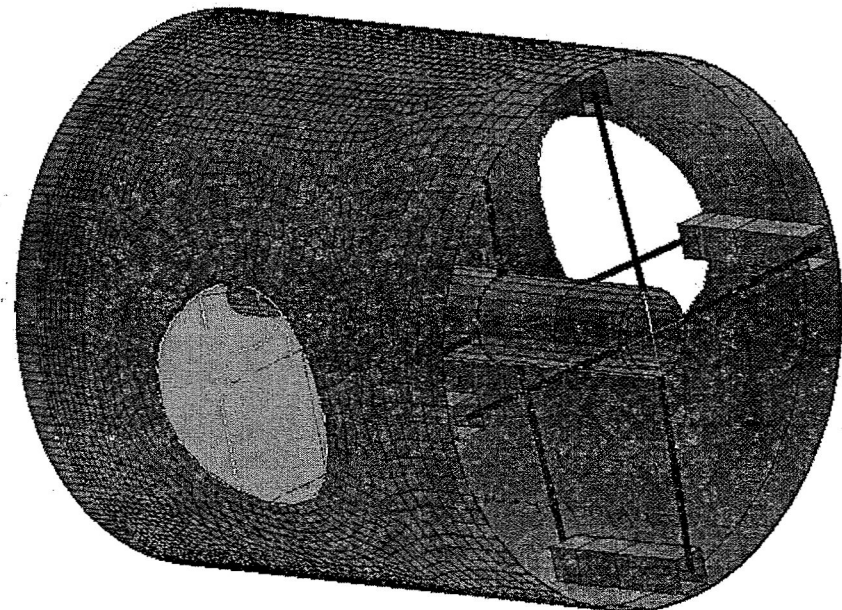
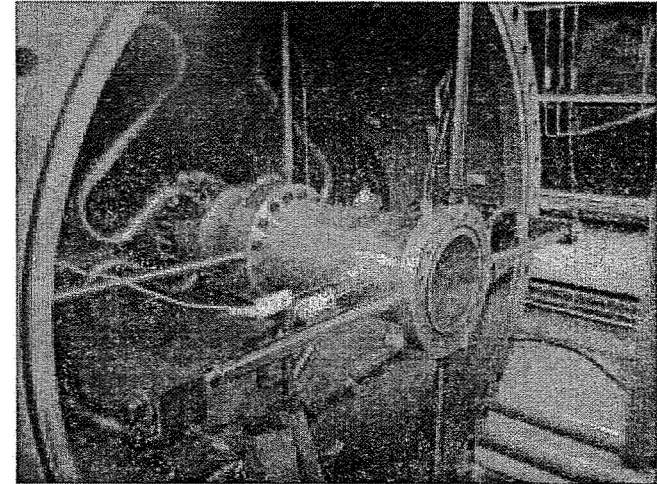
- Focuses on evaluating relative difference in side loads between several different subscale nozzle contours using cold-flow Nozzle Test Facility.
- Calculation of the side loads is performed with two methods
 - measurement of strains on a specially designed “strain-tube” that has been calibrated for static weights hung from the strain-tube nozzle interface.
 - measurement of accelerations with dynamic model to calculate effective point load that approximates net asymmetric pressure loading.





Structural Dynamic Model Generation

- first step was to generate a computer model of the nozzle and the nozzle test facility
- Initial attempts to create a finite element model of the facility could not obtain natural frequency agreement of closer than 20% for primary modes.
- modeling nozzle with a detailed finite element model and representing the boundary conditions with springs concluded to be pointless
 - boundary conditions could not possibly be represented accurately
 - no need for the detailed fe model of the nozzle only (which by itself was accurate).





“Pseudo-Model” Generation

- LMS Virtual.Lab software implements “pseudo-models” generated entirely from modal test data.
- Although this type of model offers benefits, little discussion of application of the technique is documented, aside from a recent paper by Carnes, et. al.
- Basis of technique fairly simple as applied in NASTRAN.
 - mass, spring, and damper scalar elements are connected from ground to “spoints” to simulate each natural frequency
 - spoints represent the generalized degree's of freedom $\{q\}$ in the standard modal transformation.

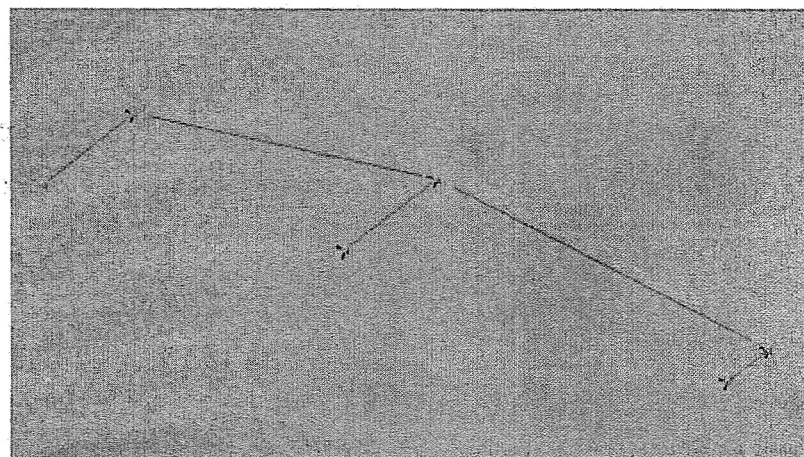
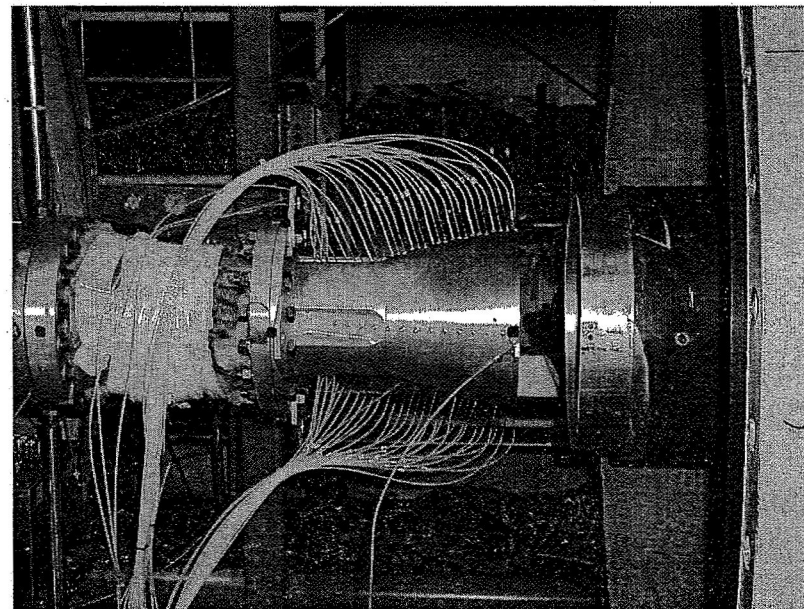
$$\{x\} = [\Phi]\{q\}$$

- Dynamic response of physical dof's expressed by generating MPC equation directly from each row of equation.
- The “pseudo-model” generated in this manner is accurate dynamically, and is created without the necessity of creating a geometry based F.E. model.



Implementation

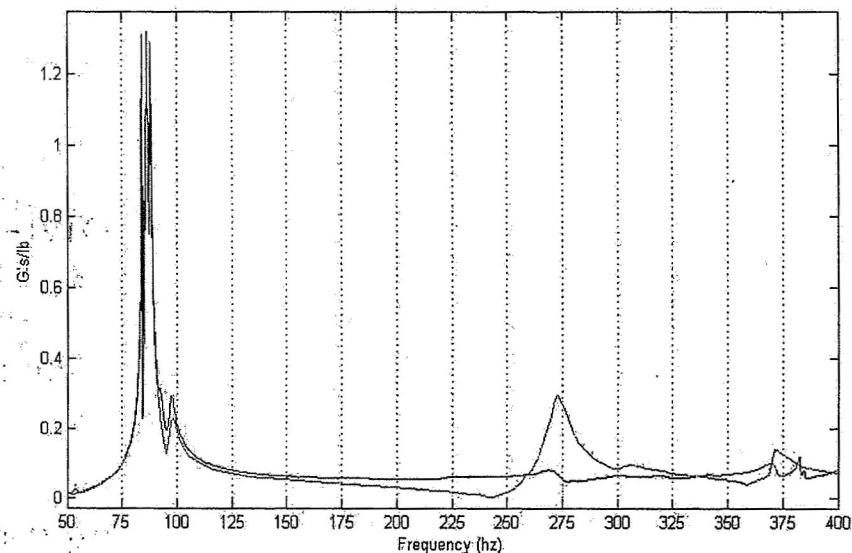
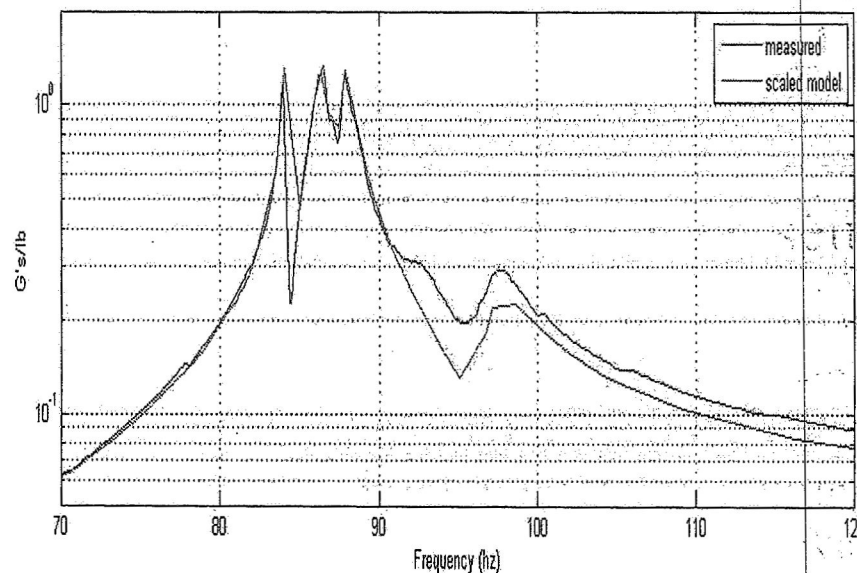
- Advantage - small number of physical points require testing, since only the dynamic response of few points may be of interest.
- ($\# \text{ tested dof's} \geq \# \text{ modes}$) for modal independence.
- Here, modal test performed with large $\#$ of dof's originally intended on being basis of pseudo-model.
- Later though, when correlation with shaker test was necessary, abbreviated 6 node modal test.
- Changes in facility configuration caused changes at fundamental system frequencies, later test chosen for pseudo-model instead.
- Subset of 16 independent modes (using MAC) selected from FRF curve fits to keep $\#$ of modes < 18 dof's.





Calibration of Pseudo-Model Response

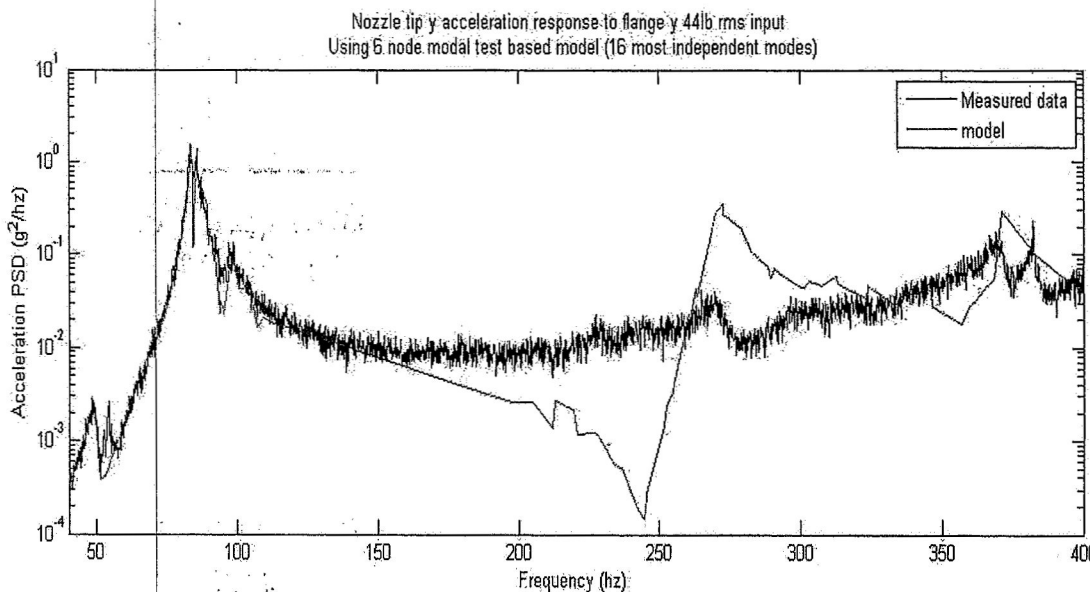
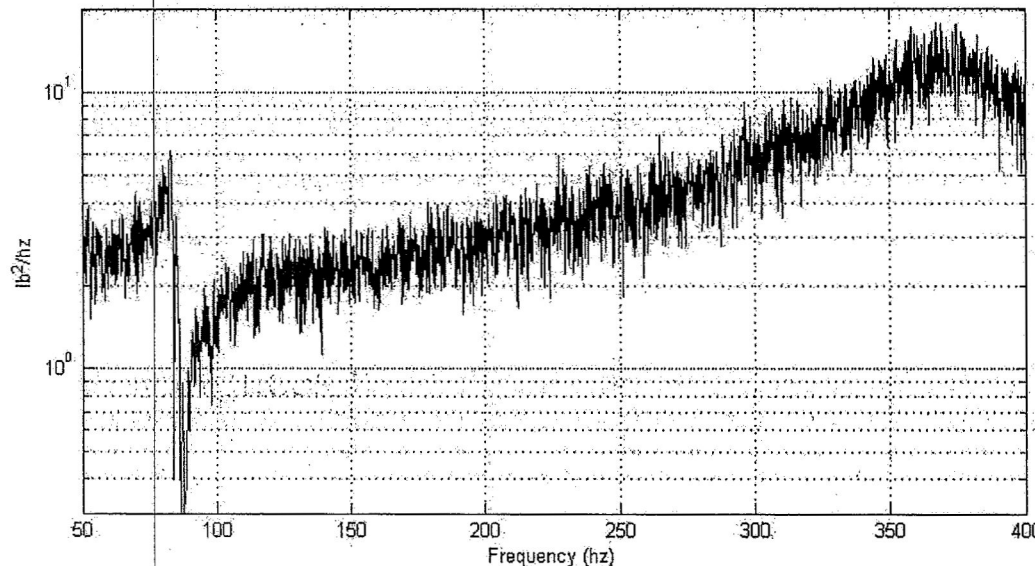
- Shakers attached to structure at flange, accelerations measured at nozzle tip.
- PSD of input force applied to pseudo-model and output FRF's compared with measured FRF's.
- Discrepancy found, reduced by imposing a factor of 0.553 on input load. Factor determined by matching FRF peak levels at fundamental frequency family ~ 78 hz.
- Resulting model FRF matches in there fairly well, but in error ~ 12th mode at 272 hz.
- Cause of error could be due to differences in exact point of load application, off-axis excitation, or ?.





Pseudo-Model Response Verification

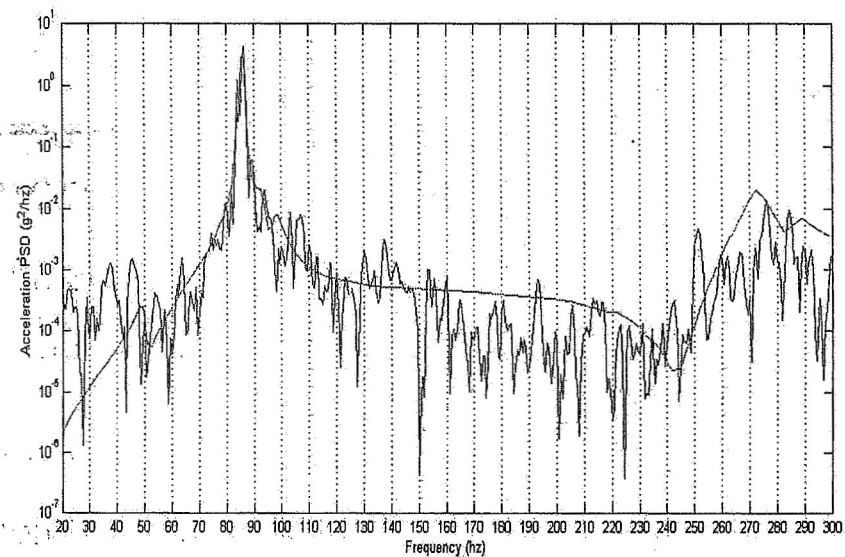
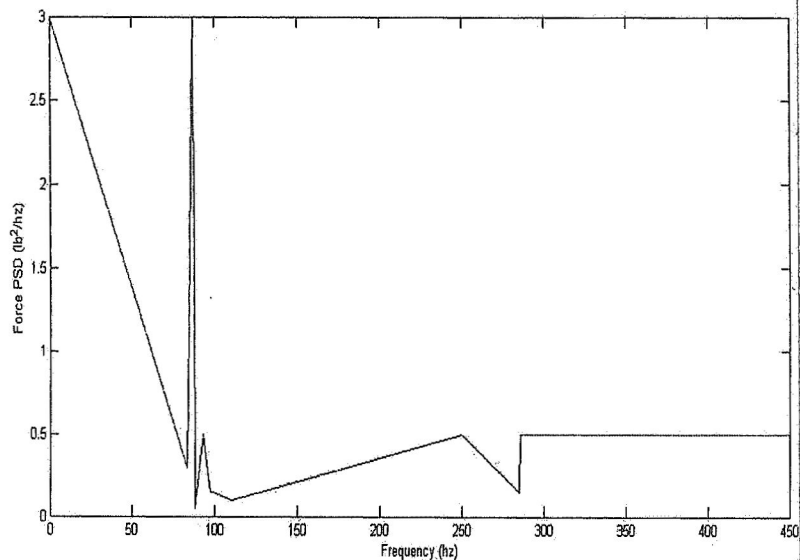
- Accel PSD responses from shaker tests then compared.
- Same results
- Same procedure was followed for lateral response and excitation – similar results, slightly different scale factor.





Inverse Force Determination

- Goal of methodology - provide a magnitude of an equivalent net force at single location.
- For convenience, location of force chosen to be the strain tube/nozzle interface.
- Initially, up-to-date method for determining this value attempted using singular value decomposition of the inverse of the FRF matrix using LMS software; didn't work.
- Decided exact methodology would take too much time to resolve, so brute-force iterative method for determining force chosen
- Assume input force frequency spectrum, apply onto ps-model, compare response PSD with measured response PSD.
- Input spectrum then modified until model and measured results matched qualitatively.
- Parameters chosen for the measured PSD were sampling rate of 20 KHz, block size of 32768, bandwidth of 0.625 hz (block of 1.6 seconds).





Results

- The RMS of the input spectrum could then be calculated and reported as the net side load.

Side Load Calculations for selected Test Points

TOC nozzle, March 1, 2006	Nozzle Pressure Ratio	RMS Side Load (lb)
367 sec	38	13.8
1144 sec	66	16.9
1009 sec	38	6.8
242 sec	unknown	12.1



Recent Work

- Realized that since assuming single input, problems with having to invert incomplete H matrix for inverse force determination don't exist.
- Applying equation

$$[S_{\ddot{x}\ddot{x}}(f)] = [H^*(f)][S_{FF}(f)][H(f)]^T$$

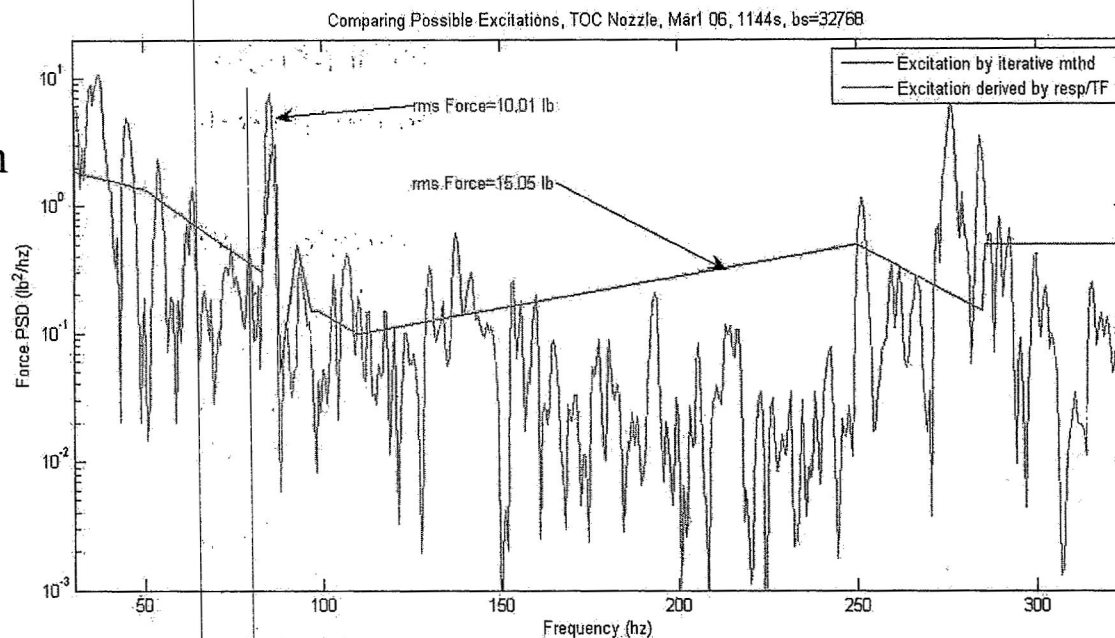
Generated “derived excitation” by simple element division of output spectra by transfer function squared.

- Although this should be the correct answer, attempted to get similar answer using pseudo-model for validation.
- Responses and input spectra now quantitatively compared using RMS calculation.
- Results do not match:
 - Transfer function in pseudo-model does not match component of H near facility mode at 270 hz
 - Due to not enough modes used in pseudo-model? – tried twice as many modes, didn't help
 - Due to pseudo-model not being well enough defined? Tried data with many more points, didn't help
- Also examined effects of stationarity of signal on block size; reduced block size in half as a result (wanted block to be stationary). Peaks are fairly well defined, but less “peaky”.

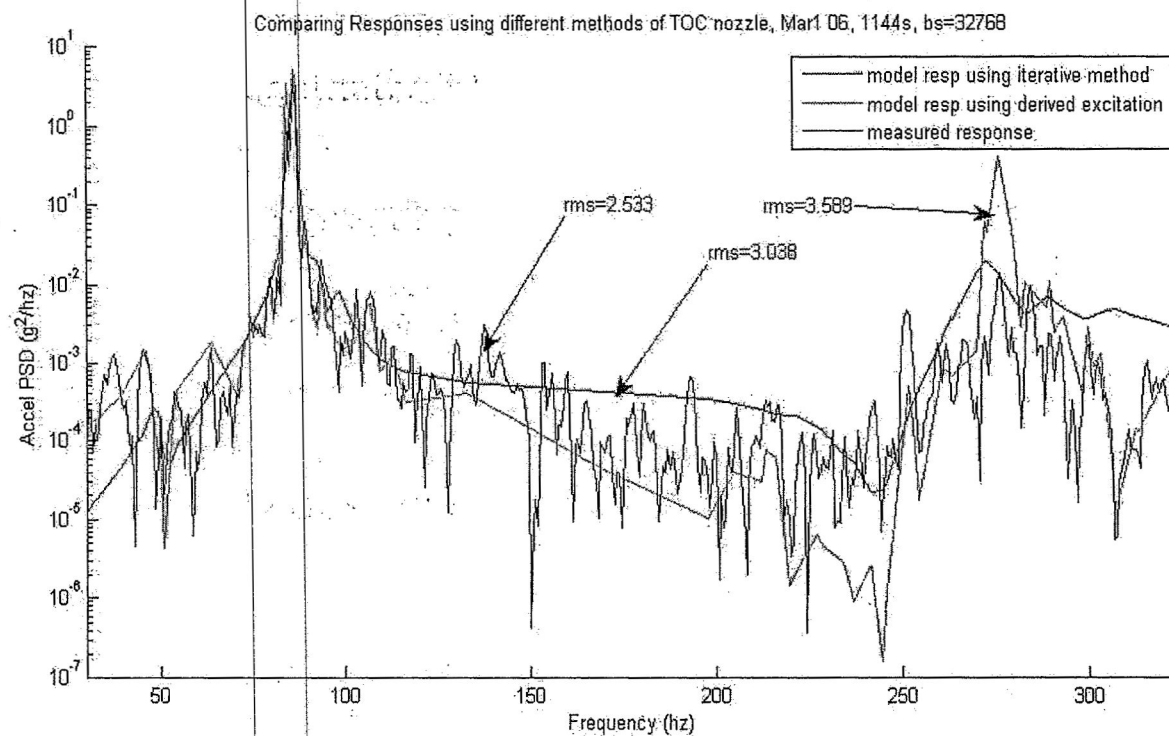


Excitation

Recent Results
(hmm...??)



Response





Summary

- Modal-test based pseudo-model used to calculate side load on sub-scale rocket nozzle using an iterative inverse load identification methodology.
 - difficult to measure integrated dynamic pressure loads inside nozzle directly.
- Although errors in the process are evident, a reasonable value of loading was obtained that can be used for design and relative comparison purposes.
- Direct calculation of loading recently performed using SISO TF relationship; results do not match ps-model iterative technique.
- Issues discussed here in using pseudo-modeling should be helpful in analyzing structures that are difficult to model and whose dynamic characteristics only are of interest.
 - Number of dof's and modes required.
 - Damping.
- Prime example of applicable structures are test-fixtures.

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NOMENCLATURE

Nozzle Test Facility	NTF
Frequency Response Function	FRF
Degree of freedom	dof
MSFC	Marshall Space Flight Center
Power Spectral Density	PSD
Root Mean Square	RMS

ABSTRACT

As part of a NASA/MSFC research program to evaluate the effect of different nozzle contours on the well-known but poorly characterized "side load" phenomena, we attempt to back out the net force on a sub-scale nozzle during cold-flow testing using acceleration measurements. Because modeling the test facility dynamics is problematic, new techniques for creating a "pseudo-model" of the facility and nozzle directly from modal test results are applied. Extensive verification procedures were undertaken, resulting in a loading scale factor necessary for agreement between test and model based frequency response functions. Side loads are then obtained by applying a wide-band random load onto the system model, obtaining nozzle response PSD's, and iterating both the amplitude and frequency of the input until a good comparison of the response with the measured response PSD for a specific time point is obtained. The final calculated loading can be used to compare different nozzle profiles for assessment during rocket engine nozzle development and as a basis for accurate design of the nozzle and engine structure to withstand these loads. The techniques applied within this procedure have extensive applicability to timely and accurate characterization of all test fixtures used for modal test.

INTRODUCTION

A major design driver for rocket engine nozzles is the evaluation of their response to the phenomena known as "side loads", which is a substantial transverse load cause by the separation of the internal flow from the side wall of the nozzle due to an overexpanded operating condition. Although this operating condition, in which the ambient pressure is greater than the internal pressure, generally only occurs at sea level ground testing of the engine, the magnitude of the transverse load has caused failures of both nozzle actuating systems [1], and sections of the nozzle itself [2]. The topic has been studied in great detail; an excellent overview of the research up to 2004 is presented by Ostland [3]. Even though the phenomena is understood on a theoretical level, though,

a technique for accurately predicting the magnitude and frequency content of the actual forcing function at a particular moment in time has not yet been determined. For this reason, researchers at NASA/MSFC have pursued a series of subscale programs in this context. The most recent investigation, led by J. Ruf, is focusing on evaluating the relative difference in side loads between several different subscale nozzle contours using the MSFC cold-flow Nozzle Test Facility (NTF). The calculation of the side loads is performed with two methods, both of which are based upon techniques applied by German researchers [4]. The first is the measurement of strains on a specially designed "strain-tube" that has been calibrated for static weights hung from the strain-tube nozzle interface (see figure 1). The second method, which is the subject of this paper, uses the measurement of accelerations with a dynamic model of the system to calculate an effective point load that approximates the net asymmetric pressure loading on the nozzle.

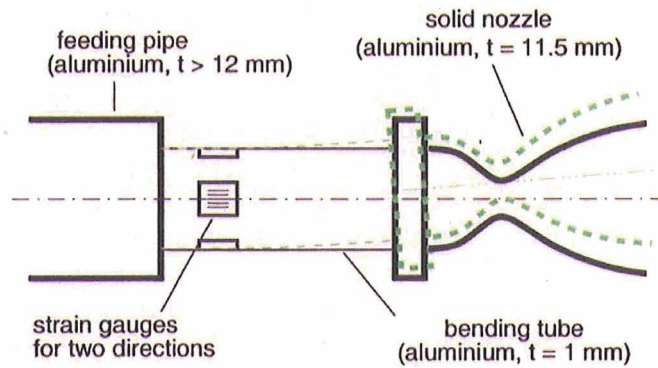


Figure 1. Schematic of Side loads test setup

MODEL GENERATION

The first step in the process was to generate a computer model of the nozzle test article and the nozzle test facility (NTF) to which it is attached (figure 2). Initial attempts to create a finite element model of the facility, a complicated structure composed of ducts, lines, stiffening rods, flow straighteners, and other poorly defined elements that have been slowly added onto for the last decade was unsuccessful; a natural frequency agreement of closer than 20% could not be achieved for the primary modes. Additional later attempts at modeling the nozzle test article only with a detailed finite element model and representing the boundary conditions accurately also were concluded to be pointless, as the representation of the enormous impact of the boundary conditions could not possibly be represented by a spring accurately and there was no need for the detailed finite element model of the nozzle test article.

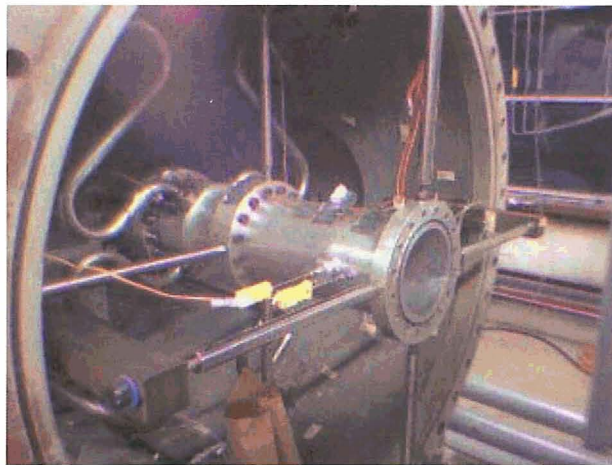


Figure 2 Photo of facility without strain tube or nozzle

At this point, a presentation on the LMS Virtual.Lab® software was made to MSFC personnel; one of the features of the software was implementation of “pseudo-models” generated entirely from modal test data. Although this type of model offers some tremendous benefits, little discussion of application of the technique is documented, aside from a recent paper by Carnes, et. al [5]. The basis of the technique is fairly simple as applied in the finite element code NASTRAN. First, mass, spring, and damper scalar elements are connected from ground to “spoints” to simulate each natural frequency, where the values taken from the parameter identification curve fit of the modal test are used for the values of modal mass, stiffness, and damping. The “spoint” is a non-physical “scalar” degree of freedom (dof) that can be referred to by multi-point constraint equations (MPC’s). In this case, the spoints represent the generalized degrees of freedom $\{q\}$ in the standard modal transformation.

$$\{x\}=[\Phi]\{q\}, \quad (1)$$

The dynamic response of the physical degrees of freedom, therefore, can be expressed by generating a MPC equation directly from each row of equation (1). The “pseudo-model” generated in this manner is accurate dynamically, and is created without the necessity of creating a geometry based finite element model.

One of the advantages of using this method is that a very small number of physical points require testing, since only the dynamic response of a few points may be of interest. The primary restriction is that the number of tested degrees of freedom equal to or exceeds the number of modes retrieved from the modal test to ensure that they can be defined independently. In this case, a modal test performed with a large number of degrees of freedom was originally intended on being the basis of the pseudo-model. Later in the process though, when correlation with a shaker test (discussed below) was necessary, an abbreviated modal test with only 6 nodes (three dof’s per node) was performed. These points are shown in red on figure 3. As some slight changes in the facility configuration were noted between the two modal tests that caused changes at the fundamental system frequencies, the abbreviated test was chosen as necessary for the pseudo-model instead. A subset of 16 modes with high values of independence as calculated using the Modal Assurance Criteria were selected from the FRF curve fits to keep the number of modes less than the number of dof’s. As with all dynamic modeling and characterization, the selection of the dof’s and modes are important in the ultimate accuracy of the results, and this probably affects the ultimate answers for this analysis as well. Enough correlation with measured results was obtained, though, as discussed below, that this pseudo-model was deemed accurate enough for this study.

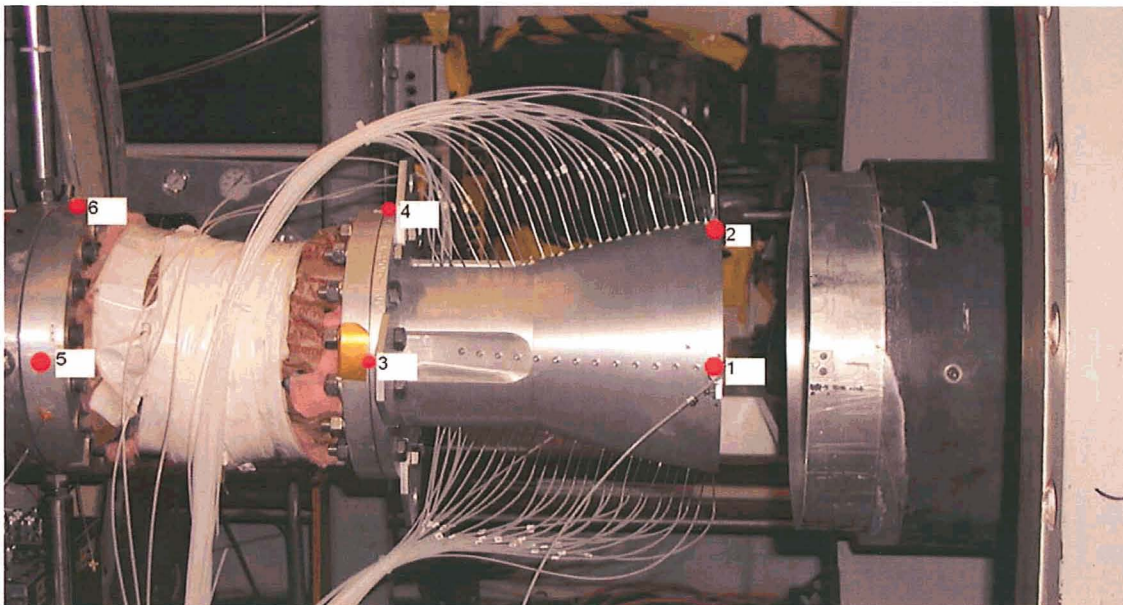


Figure 3. Strain tube (wrapped in insulation) and nozzle test article in the NTF

FRF VERIFICATION

The next step in the process was to validate that a load applied on the pseudo-model yielded the correct response. This step consisted of connecting electromagnetic shakers to the structure along the transverse axes, applying a measured input force "burst-random" spectrum, and measuring the acceleration response at the same locations as would later be measured during the actual side-loads test. The force spectrum is then applied to the pseudo-model and the response is compared with the measured response, which should be identical. Vertical and horizontal shakers were applied at points 3 and 4 in figure 3 and the response was measured at points 1 and 2. Analytically, a unit load was initially applied to the pseudo-model in the vertical direction at point 4, a frequency response analysis executed, and the FRF for the response at point 2 to this load examined. The FRF was then compared with the measured FRF. Even after resolving issues with correctly comparing test output units (G's) with analysis units, a discrepancy in the output could be seen. This discrepancy was reduced by imposing a factor of 0.553 on the input load. The factor was determined by comparing the peak levels of the FRF at the fundamental frequency family around 78 Hz. The cause of this error is not known, but could be due to differences in the exact point of load application on the test article versus the application point in the pseudo-model, off-axis excitation, or some other cause, such as unnecessarily repetitive unit conversion routines within the LMS software which reads and write the test data and the pseudo-model information. The resulting pseudo-model FRF matches the test article FRF in the fundamental frequency region fairly well (figure 4), but it is in error at the region around the 12th mode at 272 Hz (figure 5). Further investigation is planned to evaluate the source of these errors.

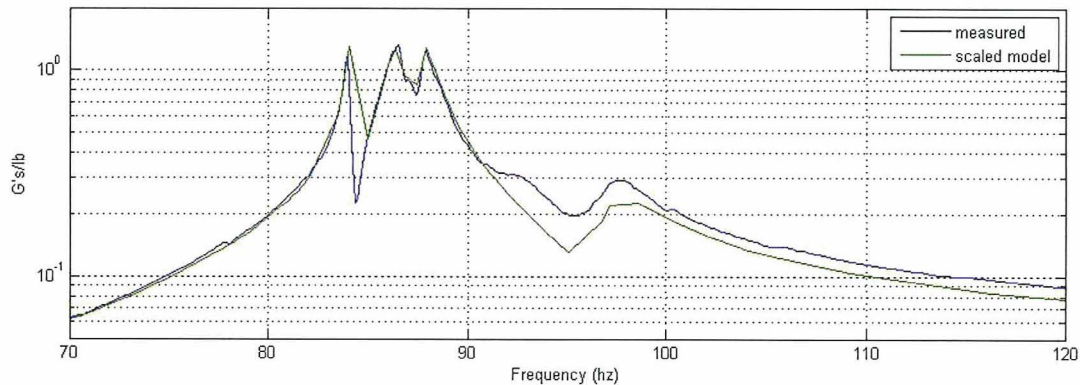


Figure 4. Flange vertical input tip vertical response FRF scaled model vs. measured comparison

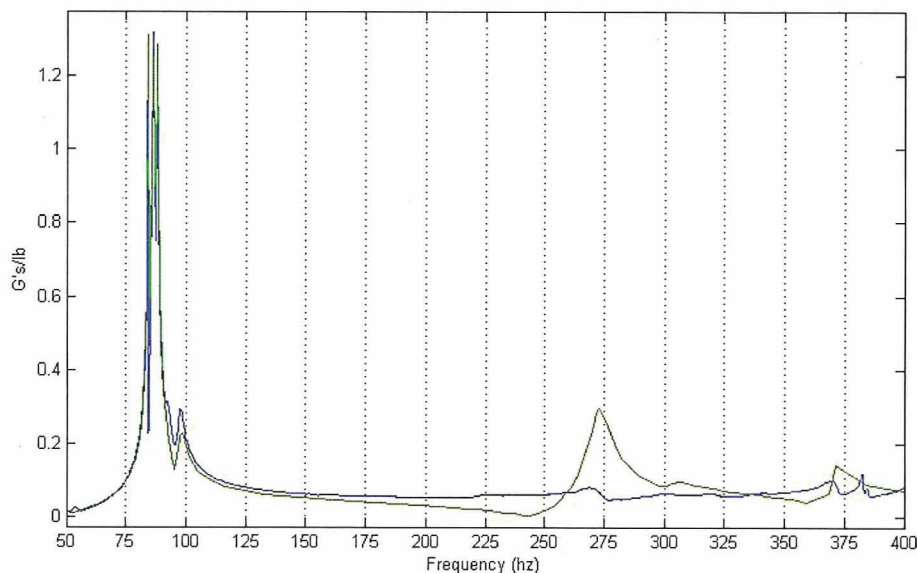


Figure 5. Overall FRF comparison

The second step in the verification process was using the same procedure for comparing the random response of the pseudo-model to the actual structural response for the shaker tests. For the measured input force spectrum (figure 6) applied by the shaker at the flange, the comparison of the pseudo-model and measured responses are shown in figure 7 and 8. As expected from the FRF results, the measured response matches well in the region of the fundamental system frequencies but is a bit off at the 272 Hz mode.

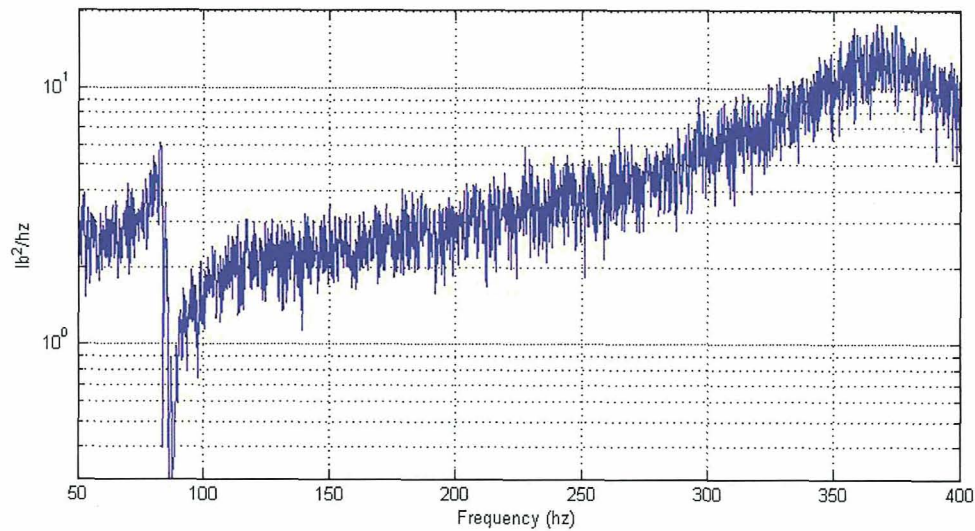


Figure 6. Shaker test vertical force input spectrum

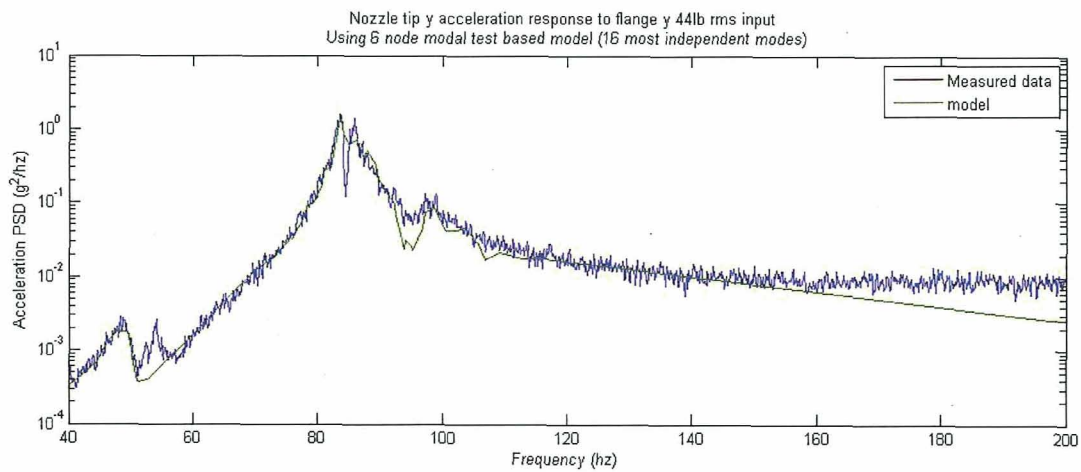


Figure 7 Shaker test response comparison

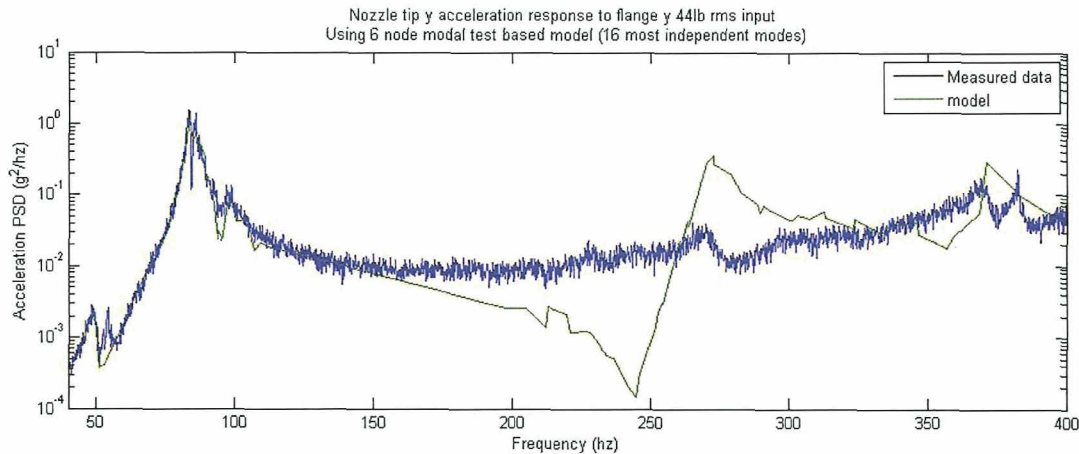


Figure 8 Shaker test response comparison overall

For verification, the same procedure was followed for lateral response and excitation. Similar results are obtained, although a slightly different load scale factor was required. This did give some additional confidence that a given level of input on either the pseudo-model or the test article would give generally the same acceleration response, so the next step in the procedure, inverse force determination, could now be pursued.

INVERSE FORCE DETERMINATION

The goal of the methodology was to provide an accurate magnitude of an equivalent single force, applied at a given point on the test facility and nozzle system, that would induce the response that was measured during side load testing. For convenience, the location of this force was chosen to be the strain tube nozzle interface (points 3 and 4), as those were the points that the shaker were attached to as well as the points where the loads for the static methodology for obtaining the side load were applied. In practice, the load is the result of an overall time and spatially varying pressure field over the inside of the nozzle, but the peak resolved load obtained using this assumption can be used for both configuration comparison and design.

Initially, an up-to-date method for determining the side load value using singular value decomposition of the inverse of the FRF matrix was attempted using the "Inverse Force Determination" module within LMS Virtual Lab. As this module was unable to read the test measured power spectral density (PSD) data, though, it made accurate determination of the input PSD very difficult. Therefore, a "brute-force" iterative method for determining the force was chosen to ensure that results would be obtained in a timely manner. This method involved assuming an input frequency spectrum, applying this loading onto the pseudo-model using MSC/Patran® and NX/Nastran®, and comparing the response PSD with the measured response PSD using Matlab®. The input spectrum was then altered based on this comparison until the model and measured results matched qualitatively. A comparison of the measured PSD at one time point with the model response PSD is shown in Figure 9 and the model input PSD generating this response is shown in figure 10. Parameters chosen for the measured PSD were a Nyquist frequency of 10,240 Hz (sampling rate of 20 KHz), block size of 32768, and a bandwidth of 0.625 Hz, which yields a block length of 1.6 seconds. Different parameter choices would yield slightly different PSD shapes, but this was the smallest block size that yielded a generally smooth PSD. The root mean square (RMS) of the input spectrum could then be calculated and reported as the net side load.

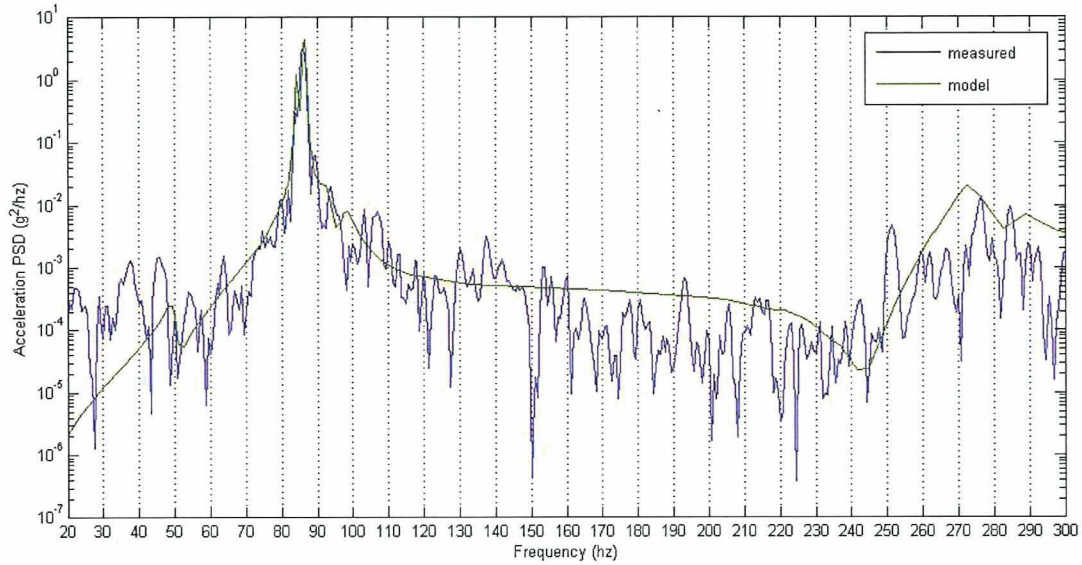


Figure 9. Comparison model, measured nozzle tip response PSD's for test on Mar1, 06, 1144sec

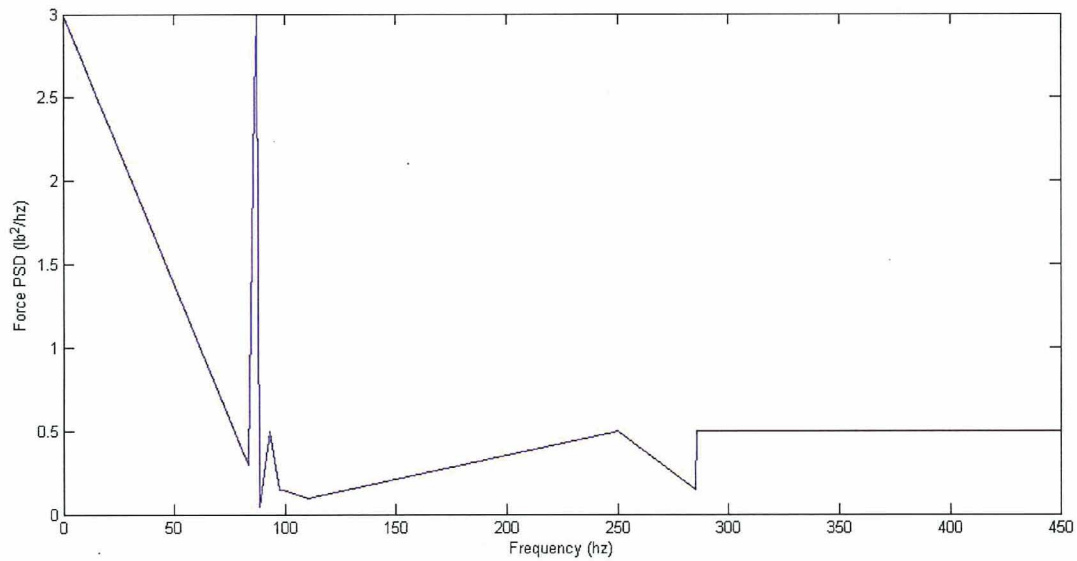


Figure 10. Iteratively generated input force PSD

Although iterative and somewhat tedious, the ability to quickly transfer information between the various software programs using modern I/O capabilities allowed the methodology to be used in a fairly repeatable and timely fashion. Once the methodology was arrived at, side loads were generated for several test conditions chosen according to maximum side load measurements obtained using the static method and for different pressure field conditions. These results are shown in table 1.

Table 1. Side Load Calculations for selected Test Points

Thrust Optimized nozzle contour, March 1, 2006	Nozzle Pressure Ratio (chamber pressure/ ambient pressure)	RMS Side Load (lb)
367 sec	38	13.8
1144 sec	66	16.9
1009 sec	38	6.8
242 sec	unknown	12.1

CONCLUSION

A modal-test based pseudo-model was used to back calculate side load on a sub-scale rocket nozzle using an iterative inverse load identification methodology. Although many errors in the process are evident, a reasonable value of loading was obtained that can be used for design and relative comparison purposes. The methodology chosen resulted from the inadequacy of existing methods for dealing with the test configuration and limitations in measurement capability inside the nozzle itself. In addition to providing useful data for the determination of rocket nozzle side loads, the limitations noted here of both the pseudo-modeling and the iterative inverse force determination method, along with their advantages, should be helpful for research and development studies of similar structures that are difficult to model and whose dynamic characteristics only are of interest. As most test-fixtures fall into this category, implementing and refining the techniques investigated in this paper should prove to be of great value in quickly and accurately assessing the dynamics of these structures.

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